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CALCULATING OPTIMUM GEAR RATIOS OF MECHANICAL DRIVEN SYSTEMS USING THREE STEP BEVEL HELICAL GEARBOX AND CHAIN DRIVE FOR MINIMUM SYSTEM HEIGHT

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ABSTRACT

The current study is aimed at determining optimum gear ratios of mechanical driven systems using a three-step bevel helical gearbox and a chain drive. An optimization problem with the objective function as the minimum gearbox height was constructed. Six input parameters including the total system ratio, the face width coefficients of the bevel and the helical gear sets, the allowable contact stress and the output torque were discussed, and their effects on the optimum gear ratios were considered through the implementation of a simulation experiment. The findings revealed, the evaluation of the factor influences, moreover, the proposed equations indicated that the optimum gear ratios can be obtained with high precision.

KEYWORDS: ***Gearbox; Bevel Helical Gearbox; Gearbox Design; Gear Ratio & Optimum Gear Ratio***

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Original Article

1. INTRODUCTION

Mechanical drive systems are popular in different fields of industry. They are used to transfer motion and torque to axes. Additionally, their size, mass and cost depend to a large extent on the transmission ratio of the system drives. Therefore, calculating the optimal transmission ratio of all levels of the mechanical drive system has attracted great attention from scientists.

So far, the gear ratios of mechanical drives along with the gearboxes have been generally calculated for different gearboxes. They are defined for helical gearbox [1–19], bevel gearbox [1, 3, 20, 21, 22], and worm gearbox [23–26]. Furthermore, they have been identified for gearboxes with different steps such as 2-step [1–8], 3-step [9–15] and 4-step [16–19] gearboxes. In addition, the studies on the determination of optimal transmission ratio have also been performed with different objective functions such as the minimum gearbox cross section area [9, 12] the minimum gearbox length [7, 11, 25], or the minimum mass of gears [10, 13, 14, 16, 18]. Recently, several studies have been carried out on obtaining gear ratios of mechanical driven systems using a gearbox and V-belt [5, 22, 27, 28, 29] or chain drive [30, 31].

This study focuses on calculating the optimum gear ratios of mechanical drive systems, using a three-step bevel helical gearbox and a chain drive to achieve the smallest gearbox height.

2. METHODOLOGY

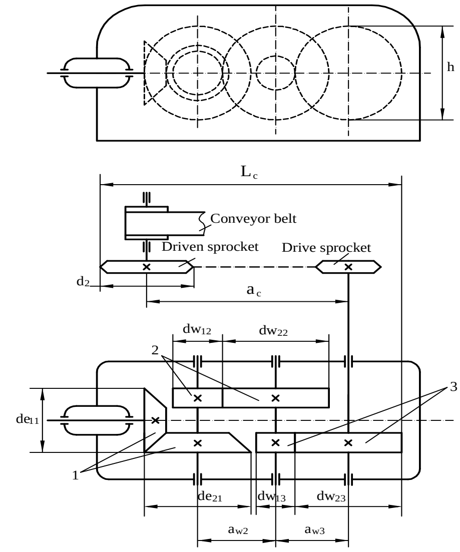

Figure 1: Calculation Schema.

The height of a mechanical driven system using a three step bevel helical gearbox and a chain drive (figure 1) can be calculated as:

$$h = \max\left(d_{e21}, d_{w22}, d_{w23}, d_2,\right) \tag{1}$$

Where, d_{e21}, d_{w22} and d_{w23} are the pitch diameters of the first, the second and the third steps, respectively; d_2 is the driven sprocket diameter of the chain drive.

As a result, the optimization problem can be written as:

$$\text{minimize } h \tag{2}$$

With several following constraints:

$$\begin{gathered} 1 \le u_1 \le 6 \\ 1 \le u_2 \le 9 \\ 1 \le u_3 \le 9 \\ 1 \le u_c \le 6 \\ u_1 \cdot u_2 \cdot u_3 \cdot u_c = u_t \end{gathered} \tag{3}$$

Where, u_1, u_2 and u_3 are the gear ratios of the first, the second, the third steps, respectively; u_c and u_t are the gear ratios of the chain drive and the total gear ratio of the system.

The above equations reveal that the requirement for solving the optimization problem is to determine d_{e21}, d_{w22}, d_{w23} and d_2.

2.1 Determining Driven Sprocket Diameter d_2

With the chain drive unit, the pitch diameter of the driven sprocket can be expressed as [32]:

$$d_2 = d_1 \cdot u_c \tag{4}$$

In which,d_1 is the pitch diameter of the drive sprocket which is determined by[32]:

$$d_1 = p / \sin(\pi / z_1) \tag{5}$$

Where, $z_1$is the number of teeth of the drive sprocket; $z_1$can be found by the following regression equation [30]:

$$z_1 = 32.4 - 2.4 \cdot u_c \tag{6}$$

P is the chain pitch (mm) determined based on the value of the design power capacity P which is considered as[32]:

$$P = P_1 \cdot k \cdot k_z \cdot k_n \tag{7}$$

In which, P_1 is the chain drive power rating (kW) which can be determined by:

$$P_1 = T_1 \cdot n_1 / (9.55 \cdot 10^6) \tag{8}$$

Where in, n_1 is the drive sprocket revolution (rpm):

$$n_1 = n_m / u_g \tag{9}$$

$$T_1 = T_{out} / (u_c \cdot \eta_c \cdot \eta_b) \tag{10}$$

In which, η_c =0.95÷0.97 is the chain drive efficiency [32]); η_b =0.99÷0.995 is the bearing efficiency [32]); T_1 and T_{out} are the drive and the output torques (Nmm). Choosing $\eta_c = 0.96$, $\eta_b = 0.992$ and substituting these values into (10) gets:

$$T_1 = 1.0502 \cdot T_{out} / u_c \tag{11}$$

k, k_z and k_n are coefficients which can be found by [32]:

$$k = k_d \cdot k_p \cdot k_c \cdot k_{adj} \cdot k_{\text{lub}} \cdot k_{con} \tag{12}$$

$$k_z = 25 / z_1 \tag{13}$$

$$k_n = n_{01} / n_1 \tag{14}$$

In equations (12) (13) and (14), the coefficients k_d, k_p, k_c, k_{adj}, k_{lub} and k_{con} reflect the effect of the shock factor, the drive position, the center distance, the adjusted possibility of the center distance, the lubrication conditions, respectively; n_{01} is the tabulated drive sprocket teeth.

2.2 Determining External Cone Distance of Bevel Gear Unit

The external cone distance R_e of the straight bevel gear unit is found by [32]:

$$R_e = k_R \cdot \sqrt{u_1^2+1} \cdot \sqrt[3]{T_{11} \cdot k_{H\beta 1} / \left[(1-k_{be}) \cdot k_{be} \cdot u_1 \cdot [\sigma_H]^2 \right]} \tag{15}$$

Where, $k_R = 50$ (Mpa$^{1/3}$) is the material coefficient; [32]; $k_{be} = 0.25 \ldots 0.3$ is the face width coefficient; [32]; $[\sigma_{H1}]$ is the allowable contact stress (MPa); $K_{H\beta 1}$ is the contact load ratio for pitting resistance determined by the regression equation found from the data in [32]:

$$K_{H\beta 1} = 0.25 \cdot k^2 + 0.2 \cdot k + 1.02 \tag{16}$$

In which, $k = k_{be} \cdot u_1 / (2 - k_{be})$.

T_{11} is the pinion torque which can be calculated by:

$$T_{11} = T_{out} / \left(u_g \cdot u_c \cdot \eta_{bg} \cdot \eta_{hg}^2 \cdot \eta_c \cdot \eta_b^4 \right) \tag{17}$$

In the above equation, T_{out} is the output torque (Nmm); u_g is the total gearbox ratio; $\eta_{hg} = 0.95 \ldots 0.97$ is the bevel gear efficiency [32]; $\eta_{hg} = 0.96 \ldots 0.98$ is the helical gear efficiency [32]; $\eta_c = 0.95 \ldots 0.97$ is the chain drive efficiency; $\eta_b = 0.99 \ldots 0.995$ is the rolling bearing efficiency [32]. Choosing $\eta_{bg} = 0.96$, $\eta_{hg} = 0.97$, $\eta_c = 0.96$ $\eta_b = 0.992$ and substituting these values into equation (17) gives:

$$T_{11} = 1.190 \cdot T_{out} / \left(u_g \cdot u_c \right) \tag{18}$$

Substituting $k_R = 50$ and (18) into (15) gets:

$$R_e = 52.984 \cdot \sqrt{u_1^2 + 1} \cdot \sqrt[3]{T_{out} \cdot \frac{k_{H\beta 1}}{[(1-k_{be}) \cdot k_{be} \cdot u_1 \cdot u_g \cdot u_c \cdot [\sigma_H]^2]}} \tag{19}$$

The outer pitch diameter of the pinion of the bevel gear set can be calculated by [32]:

$$d_{e11} = 2 \cdot R_e / \sqrt{1 + u_1^2} \tag{20}$$

And the pitch diameters of the bevel gear unit d_{e21} is determined by:

$$d_{e21} = u_1 \cdot d_{e11} \tag{21}$$

2.3 Determining the Center Distance of the Second Step

For the second step, the center distance a_{w2} is determined by [32]:

$$a_{w2} = k_m \cdot (u_2 + 1) \cdot \sqrt[3]{T_{12} \cdot k_{H\beta} / \left([\sigma_H]^2 \cdot u_2 \cdot \psi_{ba2} \right)} \tag{22}$$

Where, $k_{H\beta} = 1.02 \div 1.28$ is the contact load ratio for pitting resistance [32]; $[\sigma_H]$ is the allowable contact stress (MPa); In practice, $[\sigma_H] = 360 \ldots 420$ (MPa); k_m is the coefficient of material; k_m =43 [32]; $\psi_{ba2} = 0.3 \ldots 0.35$ is the coefficient of wheel face width [32]; T_{12} is the pinion torque which is calculated by:

$$T_{12} = T_{out}/\left(u_2 \cdot u_3 \cdot u_c \cdot \eta_{hg}^2 \cdot \eta_c \cdot \eta_b^3\right) \tag{23}$$

Choosing $\eta_{hg} = 0.97$, $\eta_c = 0.96$, $\eta_b = 0.992$ and substituting them into equation (23) gets:

$$T_{12} = 1.134 \cdot T_{out}/(u_2 \cdot u_3 \cdot u_c) \tag{24}$$

Substituting $k_{H\beta} = 1.1$, k_m =43, $k_{H\beta} = 1.1$, $u_2 \cdot u_3 = u_g / u_1$ and (24) into (22) gives:

$$a_{w2} = 46.2882(u_2 + 1) \cdot \sqrt[3]{\frac{T_{out} \cdot u_1}{[\sigma_H]^2 \cdot u_g \cdot u_2 \cdot u_c \cdot \Psi_{ba2}}} \tag{25}$$

In addition, for the second helical gear step, d_{w22} can be found by [32]:

$$d_{w22} = 2 \cdot a_{w2} \cdot u_2 / \left(u_2 + 1\right) \tag{26}$$

2.4 Determining the Center Distance of the Third Step

Similar to subsection 2.3, the center distance of the third step a_{w3} is found by [32]:

$$a_{w3} = K_m \cdot (u_3 + 1) \cdot \left[T_{13} \cdot k_{H\beta} / \left([\sigma_H]^2 \cdot u_3 \cdot \psi_{ba3}\right)\right]^{1/3} \tag{27}$$

Where, T_{13} is the output torque which is determined by:

$$T_{13} = T_{out}/\left(u_3 \cdot u_c \cdot \eta_{hg} \cdot \eta_c \cdot \eta_b^2\right) \tag{28}$$

Choosing $\eta_{hg} = 0.97$, $\eta_c = 0.96$, $\eta_b = 0.992$ as in subsection 2.3gives:

$$T_{13} = 1.0912 \cdot T_{out}/(u_3 \cdot u_c) \tag{29}$$

Substituting (29), k_m =43 and $k_{H\beta} = 1.1$ (as in section 2.3) into (27) gets:

$$a_{w3} = 45.6983 \cdot (u_3 + 1) \cdot \sqrt[3]{\frac{T_{out}}{[\sigma_H]^2 \cdot u_3^2 \cdot u_c \cdot \psi_{ba3}}} \tag{30}$$

As a result, the pitch diameter of the third step can be found by [32]:

$$d_{w23} = 2 \cdot a_{w3} \cdot u_3 / \left(u_3 + 1\right) \tag{31}$$

Experimental Work

In this study, to investigate the effect of input parameters on optimal partial gear ratios, a simulation experiment was designed and implemented. In the experiment, a 2-level full factorial design and 6 input parameters were selected

(table 1). Thus, the experiment was designed with $2^6 = 64$ numbers of tests. In addition, to perform the experiment, a computer program was built based on Equations (2) and (3). The different levels of the input factors and the output responses (the optimum gear ratios of the second and the third step (u_2 and u_3) and the chain drive u_c are presented in table 2.

Table 1: Input Parameters

Factor	Code	Unit	Low	High
Total gearbox ratio	u_t	-	40	70
Coefficient of the face width of bevel gear set	Kbe	-	0.25	0.3
Coefficient of wheel face width of step 2	x_{ba2}	-	0.3	0.35
Coefficient of wheel face width of step 3	x_{ba3}	-	0.35	0.4
Allowable contact stress	AS	MPa	350	420
Output Torque	T_{out}	Nm	10^2	10^4

3. RESULTS AND DISCUSSIONS

3.1 Influences of Input Parameters on the Optimum Gear Ratio of the Chain Drive

To evaluate the influences of the input parameters on the optimum gear ratio of the chain drive u_c, figure 2 describes the graph of the main effects for u_c. It is noticeable from the graph that u_c is greatly influenced by the total system gear ratio. In addition, it depends to a lesser extent on ψ_{ba2}, ψ_{ba3} and k_{be} in descending order. Also, it is not determined by the coefficient of the face width of bevel gear set k_{be} and the allowable contact stress AS.

Table 2: Experimental Plans and Output Response

Run Order	Center Pt	Blocks	u_t	K_{be}	Xba1	Xba2	AS (MPa)	Tout (Nm)	u_c	u_2	u_3
1	1	1	40	0.25	0.3	0.35	350	10000	4.20	4.35	2.27
2	1	1	40	0.3	0.35	0.4	350	100	4.00	4.11	2.27
3	1	1	70	0.25	0.3	0.4	420	10000	5.60	5.60	2.60
4	1	1	70	0.25	0.3	0.4	350	100	5.60	5.27	2.60
5	1	1	70	0.25	0.35	0.4	420	10000	5.70	5.87	2.46
6	1	1	40	0.25	0.35	0.4	420	10000	3.90	4.32	2.33
63	1	1	70	0.3	0.35	0.35	420	100	5.80	5.81	2.32
64	1	1	70	0.3	0.35	0.4	420	10000	5.70	5.87	2.46

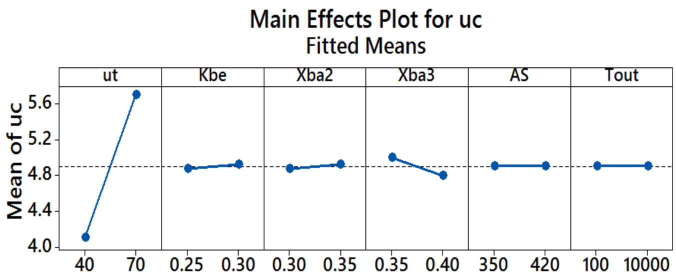

Figure 2: Main Effects Plot for u_c.

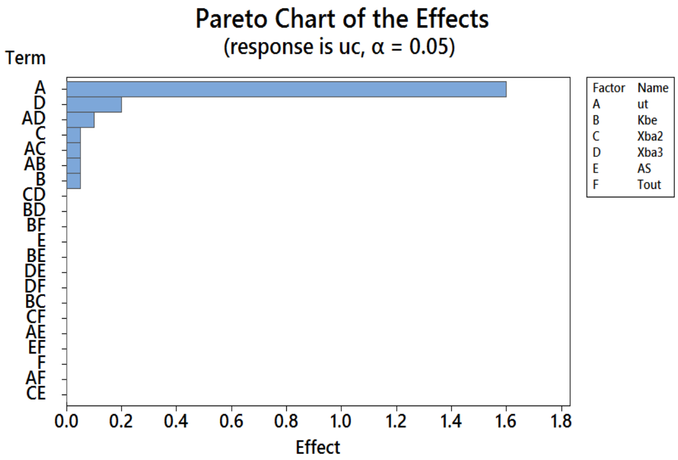

Figure 3: Pareto Chart for u_c.

Coded Coefficients

Term	Effect	Coef	SE Coef	T-Value	P-Value	VIF
Constant		4.900	0.000	*	*	
ut	1.6000	0.8000	0.0000	*	*	1.00
Kbe	0.05000	0.02500	0.00000	*	*	1.00
Xba2	0.05000	0.02500	0.00000	*	*	1.00
Xba3	-0.2000	-0.1000	0.0000	*	*	1.00
ut*Kbe	-0.05000	-0.02500	0.00000	*	*	1.00
ut*Xba2	0.05000	0.02500	0.00000	*	*	1.00
ut*Xba3	0.10000	0.05000	0.00000	*	*	1.00

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0	100.00%	100.00%	100.00%

Figure 4: Estimated Effects and Coefficients for u_c.

The Pareto chart of the standardized effects for u_c is demonstrated in figure 3. From this figure, it can be learned that the bars that represent input parameters including the total gearbox ratio (factor A), the coefficients of wheel face width of the third (factor D), the second (factor C) and the first (factor B) gear units and the interactions AD, AC and AB cross the reference line (with value 0 on the x-axis). Consequently, these factors are statistically significant at the 0.05 level with u_c.

Figure 4 shows the estimated effects and coefficients for u_c. It is found that u_t, k_{be}, ψ_{ba2}, ψ_{ba3} and the interactions $u_t \cdot k_{be}, u_t \cdot \psi_{ba2}$ and $u_t \cdot \psi_{ba3}$ are significant affecting factors on u_c. Hence, the relation between u_c and these factors are determined by:

$$u_c = 5.8 - 0.00000001 \cdot u_t + 4.667 \cdot k_{be} - 2.667 \cdot \psi_{ba2} - 11.33 \cdot \psi_{ba3} - \\ -0.06667 \cdot u_t \cdot k_{be} + 0.06667 \cdot u_t \cdot \psi_{ba2} + 0.1333 \cdot u_t \cdot \psi_{ba3} \quad (32)$$

The values of adj-R2 and pred-R2 are 100% (figure 4), which indicates that Equation (32) matches the experiment data..

3.2 Effect of Input Parameters on the Optimum Gear Ratio of the Second Step

It is interesting that the optimum gear ratio of the first step of the gearbox does not depend on the input parameters and:

$$u_1 = 1 \tag{33}$$

That means, with the objective of the optimization problem, the bevel gear unit is only capable of transmitting torque without executing the deceleration task.

3.3 Effect of Input Parameters on the Optimum Gear Ratio of the Third Step

The effects of the input parameters on the optimum gear ratio of the third step u_3 are presented in figure 5. It can be seen from the figure that the largest effect belongs to u_t, followed by ψ_{ba2}, ψ_{ba3}, T_{out} and AS. Besides, k_{be} does not effect on u_3 as the slope of its graph is 0.

The Normal Plot of the standardized effects is described in figure 6. From this graph, it can be realized that u_2 largely depends on the system gear ratio u_t. However, it slightly depends on ψ_{ba2}, ψ_{ba3}, T_{out}, AS and the interactions $u_t \cdot \psi_{ba2}$, $u_t \cdot \psi_{ba3}$ and $u_t \cdot T_{out}$ in descending order.

Moreover, u_t (factor A), ψ_{ba2} (factor C), T_{out} (factor F) and the interactions $u_t \cdot T_{out}$ (AF) and $u_t \cdot \psi_{ba3}$ (AD) have a positive standardized effect. If these factors go up, u_3 increases. In contrast, ψ_{ba3} (factor D), AS (factor E) and the interaction $u_t \cdot \psi_{ba2}$ (AC) have a negative standardized effect. That means if these factors rise, u_3 declines.

Figure 7 demonstratesthe estimated effects and coefficients for u_2. It was reported that u_t, ψ_{ba2}, ψ_{ba3}, AS, T_{out} and the interactions $u_t \cdot \psi_{ba2}$, $u_t \cdot \psi_{ba3}$ and $u_t \cdot T_{out}$ are the significant affecting factors on u_2. Therefore, u_2 can be calculated by:

$$\begin{aligned} u_2 &= 1.679 + 0.07583 \cdot u_t - 8.833 \cdot k_{be} + 0.833 \cdot \psi_{ba2} + 5.333 \cdot \psi_{ba3} + 0.03333 \cdot u \cdot k_{be} - \\ &- 0.03333 \cdot u_t \cdot \psi_{ba2} - 0.13333 \cdot u_t \cdot \psi_{ba3} + 20 \cdot k_{be} \cdot \psi_{ba2} \end{aligned} \tag{34}$$

Equation (34) is used to find the gear ratio of the second step u_2. After gaining u_1 (Equation 33), u_2 (Equation 34)and u_c (Equation 32), the gear ratio of the third step can be expressed by:

$$u_3 = u_t / \left(u_1 \cdot u_2 \cdot u_c \right) \tag{35}$$

As $u_1 = 1$, formula (35) is rewritten as:

$$u_3 = u_t / \left(u_2 \cdot u_c \right) \tag{36}$$

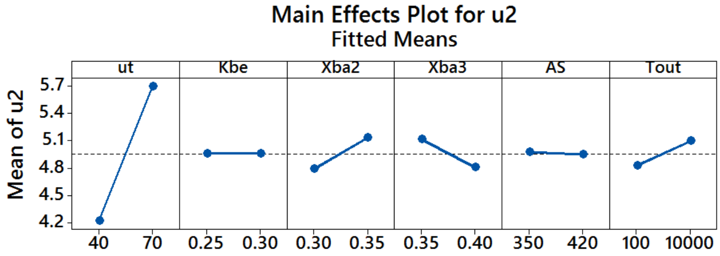

Figure 5: Main Effects Plot for Optimum Gear Ratio of u_2.

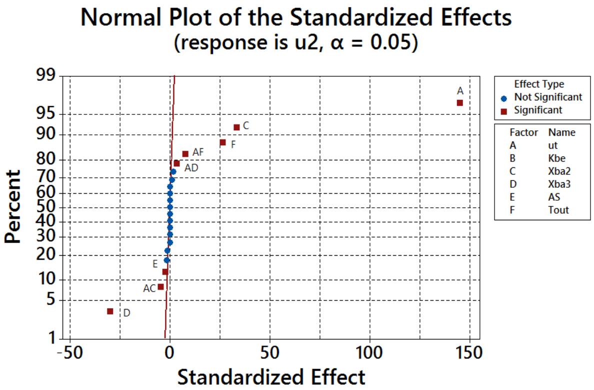

Figure 6: Normal Plot for u_2.

Coded Coefficients

Term	Effect	Coef	SE Coef	T-Value	P-Value	VIF
Constant		4.63750	0.00169	2751.41	0.000	
ut	0.72500	0.36250	0.00169	215.07	0.000	1.00
Kbe	-0.02500	-0.01250	0.00169	-7.42	0.000	1.00
Xba2	0.22500	0.11250	0.00169	66.75	0.000	1.00
Xba3	-0.10000	-0.05000	0.00169	-29.66	0.000	1.00
ut*Kbe	0.02500	0.01250	0.00169	7.42	0.000	1.00
ut*Xba2	-0.02500	-0.01250	0.00169	-7.42	0.000	1.00
ut*Xba3	-0.10000	-0.05000	0.00169	-29.66	0.000	1.00
Kbe*Xba2	0.02500	0.01250	0.00169	7.42	0.000	1.00

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.0134840	99.90%	99.88%	99.86%

Figure 7: Estimated Effects and Coefficients for u_2.

4. CONCLUSIONS

This paper introduces a study on determining the optimum gear ratios of mechanical driven systems, using a three step bevel helical gearbox and a chain drive. In the problem, the minimum system height was selected as the objective function, and six input factors including the total system gear ratio, the coefficients of the face width of three steps, the allowable contact stress and the output torque were inspected. To estimate the influences of these factors on the optimum gear ratios, a simulation experiment was designed and conducted. Considerably, several equations to calculate the optimum gear ratios were suggested.

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REFERENCES

1. *V.N. Kudreavtev; I.A. Gierzaves; E.G. Glukharev, Design and calculus of gearboxes (in Russian), Mashinostroenie Publishing, Sankt Petersburg, 1971.*

2. *Trinh Chat, 1993 Some problems of kinematics calculation of transmission mechanics system (in Vietnamese) Proc. of the National Conference on Engineering Mechanics Vol. 2 (Hanoi) pp. 7–12.*

3. *G. Milou, G. Dobre, F. Visa, H. Vitila, 1996. Optimal Design of Two Step Gear Units, regarding the Main Parameters. VDI Berichte No 1230, p. 227.*

4. *Vu Ngoc Pi, 2001. A method for optimal calculation of total transmission ratio of two step helical gearboxes. Proceedings of the National conference on Engineering Mechanics, Ha Noi, p. 12.*

5. *Nguyen Thi Hong Cam, Vu Ngoc Pi, Nguyen Khac Tuan, Le Xuan Hung, Tran Thi Phuong Thao, 2019. Determining optimal partial transmission ratios of mechanical driven systems using a V-belt drive and a helical reducer with second-step double gear-sets. H. Fujita et al. (Eds.): ICERA 2018, LNNS 63, pp. 261–269, https://doi.org/10.1007/978-3-030-04792-4_35.*

6. *A.N. Petrovski, B.A. Sapiro, N.K. Saphonova, 1987. About optimal problem for multi-step gearboxes (in Russian). Vestnik Mashinostroenie, No. 10, p. 13.*

7. *Vu Ngoc Pi, A new study on optimal calculation of partial transmission ratios of two-step helical gearboxes, 2nd WSEAS Int. Conf on Computer Engineering and Applications (CEA'08), Acapulco, Mexico, January 25–27, 2008, pp. 162–165.*

8. *Khac Tuan Nguyen, Ngoc Pi Vu, Thi Hong Cam Nguyen, Thi Phuong Thao Tran, Ky Thanh Ho, Xuan Hung Le and Thi Tham Hoang, Determining Optimal Gear Ratios of a Two-stage Helical Reducer for Getting Minimal Acreage of Cross Section, MATEC Web of Conferences 213, 01008 (2018), https://doi.org/10.1051/matecconf/201821301008*

9. *Vu Ngoc Pi, Nguyen Khac Tuan, 2016. Optimum determination of partial transmission ratios of Three-step helical gearboxes for getting minimum cross section. Journal of Environmental Science and Engineering A 5, pp. 570–573.*

10. *Shuai, M., Yidu, Z., & Qiong, W. (2015). Research on multiple-split load sharing of two-stage star gearing system in consideration of displacement compatibility. Mechanism and Machine Theory, 88, 1–15.*

11. *Vu Ngoc Pi, Nguyen Dang Binh, Vu Quy Dac, Phan Quang The, 2006. Optimal Calculation of Total Transmission Ratio of Three-Step Helical Gearboxes for Minimum Mass of Gears. (In Vietnamese) Journal of Science and Technology of 6 Engineering Universities, p. 91.*

12. *Vu Ngoc Pi, 2008. A new study on optimal calculation of partial transmission ratio of three-step helical reducers. The 3rd IASME / WSEAS International Conference on Continuum Mechanics, Cambridge, UK, p. 23.*

13. *Vu Ngoc Pi, 2008. A new study on optimal calculation of partial transmission ratios of three-step helical reducers for getting minimal cross section dimension. The 2nd WSEAS International Conference on Computer Engineering and Applications (CEA'08), Acapulco, Mexico January 25-27, pp. 290–293.*

14. *Vu Ngoc Pi, 2008. Optimal determination of partial transmission ratios of three-step helical gearboxes with first and third-step double gear-sets for minimal mass of gear. American Conference on Applied Mathematics (MATH'08), Harvard, Massachusetts, USA March 24–26, pp. 385–388.*

15. *Romhild I, Linke H, 1992. Gezielte Auslegung Von Zahnradgetrieben mit minimaler Masse auf der Basis neuer Berechnungsverfahren. Konstruktion 44, pp. 22–9236.*

16. *Vu Ngoc Pi, Nguyen Khac Tuan, Optimum Determination of Partial Transmission Ratios of Three-Step Helical Gearboxes for Getting Minimum Cross Section Dimension, Journal of Environmental Science and Engineering A 5 (2016) 570–573.*

17. *Vu Ngoc Pi, 2008. Optimal determination of partial transmission ratios for four-step helical gearboxes with firt and third step double gear-sets for minimal mass of gears. Applied Computing Conference (ACC' 08), Istanbul, Turkey, May 27–30.*

18. *Vu Ngoc Pi, 2008. Optimal calculation of partial transmission ratios for four-step helical gearboxes with first and third-step double gear-sets for minimal gearbox length. American Conference on Applied Mathematics (MATH'08), Harvard, Massachusetts, USA March 24–26, pp. 29–32.*

19. *Le Xuan Hung, Vu Ngoc Pi, Nguyen Van Du, September 2009. Optimal calculation of partial transmission ratios of four-step helical gearboxes with second and fourth-step double gear-sets for minimal mass of gears. The international symposium on Mechanical Engineering (ISME Ho Chi Minh city, Vietnam, pp. 21–23.*

20. *Ahmad, F., Choi, H. S., & Park, M. K. (2015). A review: natural fiber composites selection in view of mechanical, light weight, and economic properties. Macromolecular Materials and Engineering, 300(1), 10–24.*

21. *Vu Ngoc Pi, Optimal determination of partial transmission ratios for four-step helical gearboxes with first and third step double gear-sets for minimal mass of gears, Applied Computing Conference (ACC '08), Istanbul, Turkey, May 27–30, 2008, pp. 53–57.*

22. *Vu Ngoc Pi, A new and effective method for optimal calculation of total transmission ratio of two step bevel - helical gearboxes, International colloquium on Mechannics of Solids, Fluids, Structures & Interaction Nha Trang, Vietnam, 2000, 716–719.*

23. *Vu Ngoc Pi, Nguyen Dang Binh, Vu Quy Dac, Phan Quang The, A new and effective method for optimal splitting of total transmission ratio of three step bevel-helical gearboxes, The Sixth Vietnam Conference on Automation, Hanoi (2005), 175–180.*

24. *Nguyen Khac Tuan, Tran Thi Phuong Thao, Nguyen Thi Hong Cam, Le Xuan Hung, Vu Ngoc Pi, Optimum calculation of partial transmission ratios of mechanical driven systems using a V-belt and a three-step bevel helical gearbox, H. Fujita et al. (Eds.): ICERA 2018, LNNS 63, pp. 469–476, 2019, https://doi.org/10.1007/978-3-030-04792-4_61*

25. *Chernavsky S A et al. 1984. Design of mechanical transmissions: Manual for high technical schools (Moscow: Mashinostroenie) p 560.*

26. *Vu Ngoc Pi and Vu Quy Dac, Calculation of total transmission ratio of two step worm reducers for the best reasonable gearbox housing structure (in Vietnamese), J. of Science and Technology - Thai Nguyen University Vol. 1 (41), 2007, pp 65–69.*

27. *Vu Ngoc Pi and Vu Quy Dac, Optimal calculation of partial transmission ratios of worm-helical gear reducers for minimal gearbox length, J. of Science & Technology - Technical Universities, No. 61, 2007, pp. 73–77.*

28. *Vu Ngoc Pi, Vu Quy Dac, Optimal calculation of total transmission ratio of worm-helical gear reducers, Journal of Science and Technology, Thai Nguyen University, Vol. 4 (36), No.1, 2005, 70–73.*

29. *Trinh Chat and Le Van Uyen, 2007 Design and calculation of Mechanical Transmissions Systems Vol. 1, Hanoi, Educational Republishing House.*

30. *Vu Ngoc Pi, Tran Thi Phuong Thao, Le Thi Phuong Thao, 2015. A new study on optimum determination of partial transmission ratios of mechanical driven systems using a V-belt and two-step helical gearbox. Vietnam Mechanical Engineering Journal, No. 10, p. 123.*

31. *Vu Ngoc Pi, Nguyen Thi Hong Cam, Nguyen Khac Tuan, 2016. Optimum calculation of partial transmission ratios of mechanical driven systems using a V-belt and two-step bevel helical gearbox. Journal of Environmental Science and Engineering A 5, p. 566.*

32. *Nguyen Thi Hong Cam, Vu Ngoc Pi, Nguyen Khac Tuan, Le Xuan Hung, Tran Thi Phuong Thao, Determining optimal partial transmission ratios of mechanical driven systems using a V-belt drive and a helical reducer with second-step double gear-sets, H. Fujita et al. (Eds.): ICERA 2018, LNNS 63, pp. 261–269, 2019, https://doi.org/10.1007/978-3-030-04792-4_35*

33. *Vu Ngoc Pi, Tran Thi Phuong Thao, Dang Anh Tuan, 2017. Optimum determination of partial transmission ratios of mechanical driven systems using a chain drive and two-step helical gearbox. Journal of Environmental Science and Engineering B 6, p. 80.*

34. *Nguyen Thi Hong Cam, Vu Ngoc Pi, Nguyen Khac Tuan, Le Xuan Hung, Tran Thi Phuong Thao, A study on determination of optimum partial transmission ratios of mechanical driven systems using a chain drive and a three-step helical reducer, H. Fujita et al. (Eds.): ICERA 2018, LNNS 63, pp. 91–99, 2019, https://doi.org/10.1007/978-3-030-04792-4_14*

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